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FOR BALL AND ROLLER BEARINGS**

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HOW TO APPLY LIFE ADJUSTMENT FACTORS FOR BALL AND ROLLER BEARINGS

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ABSTRACT

Practical problems applicable to the selection, design and lubrication of rolling-element bearings are presented and discussed. The solutions to these problems are based upon the new ASME Engineering Design Guide - Life Adjustment Factors for Ball and Roller Bearings. Design and selection criteria are based upon materials and processing factors such as melting practice, metalworking, and heat treatment. Environmental factors are considered such as bearing misalignment and speed. Selection of a lubricant is based upon elastohydrodynamic lubrication principles in addition to lubricant type and chemistry.

INTRODUCTION

In recent years, mechanical engineers, specifically in the field of design, have been required to incorporate advances in the state-of-the-art when selecting rolling-element bearings. The life predictions contained in manufacturers' catalogues which are based on the Anti-Friction Bearing Manufacturers Association (AFBMA) rating are probably adequate for most applications. However, they do not reflect state-of-the-art advances in materials, processing variables, elastohydrodynamic lubrication, lubricant type, manufacturing variables, and operating speed. The Rolling-Elements Committee of the ASME Lubrication Division recognized this and published in September 1971 a design guide entitled, "Life Adjustment Factors for Ball and Roller Bearings" [1].¹ This design guide extends the "engineering approximations" which are illustrated in most bearing manufacturers' catalogues and provides information that will be of most use to the engineer.

The AFBMA method [2] for determining bearing load rating and fatigue life, and/or basic ratings as published in any bearing manufacturer's catalog must be the heart of any design guide. Continuing in the vein of

¹Brackets refer to references in back of text.

the engineering approximation, it is assumed that various environmental or bearing design factors are, at least for first-order effects, multiplicative. As a result, the expected bearing life L_A , which is based upon a fatigue criterion of failure, can be related to the calculated rating life² L_{10} by the following:

$$\begin{aligned} L_A &= (D)(E)(F)(G)(H)L_{10} \\ &= (D)(E)(F)(G)(H)\left(\frac{C}{P}\right)^n \end{aligned} \quad \begin{matrix} 3 \\ (1) \end{matrix}$$

where

D materials factor

E processing factor

F lubrication factor

G speed effect factor

H misalignment factor

C basic load rating

P equivalent load (defined in bearing manufacturer's catalog or AFBMA standards)

n load-life exponent; either 3 for ball bearings or 10/3 for roller bearings

The engineer must be cautioned, however, that in some instances the manufacturer may have incorporated life factors into his catalog ratings independent of AFBMA. Therefore, prior to using equation (1), the engineer should establish either the manufacturer's life factor or the basic AFBMA rating.

Factors D and E which deal primarily with premium steels, special material composition, or special processing techniques, normally require special manufacturing consideration. This may lead to increased bearing costs. The remainder of the factors, however, can be applied within certain reasonable limits to most ball or roller bearing applications.

It, therefore, becomes the objective of this paper to present examples of the use of the life adjustment factors as applied to typical ball and roller bearing applications. Lubrication and material considerations

²The L_A and L_{10} lives are statistical lives based upon a 90 percent probability of survival for this time period. If a bearing's mode of failure is other than by fatigue, this life prediction may not be applicable.

³Equation, table, and figure numbers refer to those equations of like number in [1].

will be applied to illustrate to the design engineer how the design guide may be used in those life-critical applications for which special bearings or design considerations are justified. The reader must, however, be cautioned that the following problems are hypothetical and are not meant to be overall encompassing in nature. As a result, the design engineer must perform his own calculations applicable to his specific problem area using [1].

Example 1

In a petrochemical processing operation, a ball bearing spindle drives two vane pumps mounted internally in two separate pipe lines. The spindle is belt driven at 600 rpm and utilizes two 209 (45-mm bore diameter) size radially loaded ball bearings. The bearings each have a complement of nine (9) one-half (1/2) inch diameter balls. Both the bearing rings and balls are made from consumable-electrode vacuum-melted AISI T-1 steel (see table 1 of [1]). The material hardness ranged from Rockwell C 58 to 62. The radial load on each bearing is 2000 pounds. Using standard AFBMA calculations [2], the basic load rating C of the bearing is 5676 pounds.

From the following equation, the L_{10} life of the bearing can be determined.

$$L_{10} = \left(\frac{C}{P} \right)^3 = \left(\frac{5676}{2000} \right)^3$$

$$= 22.8 \times 10^6 \text{ revolutions}$$

Since the bearing operates at 600 rpm, the bearing life in hours would be equal to $(22.8 \times 10^6) / (600 \times 60)$ or approximately 630 hours. Because the spindle has two bearings we must consider the spindle as a system. It must be remembered that the 630 hours is the time the bearing can theoretically achieve with a 90-percent probability of survival. Since there are two bearings, the system probability of survival at 630 hours is $(0.9)^a$ (a is the number of bearings) or 81 percent. Using the statistical methods of [3], the system life at a 90 percent probability of survival becomes $630/a$ or 315 hours. Since this is a continuous operation, the user would like to operate without failure between the semi-annual shut-down for general repair and overhaul.

A bearing failure necessitates shutting down two process lines at considerable expense. Currently, the plant is experiencing bearing failures between four weeks and six months. There are 12 pumps in use, simultaneously. If the first bearing fails in four weeks or 672 hours, this failure represents, on a first order approximation, $(1-1/24) \times 100$ or the 96 percent probability of survival. (This number is a rough approximation. For a more accurate analysis utilizing mean order numbers, see [3].) This means that the actual bearing L_{10} life in the application is approximately $[(1-0.90)/(1-0.96)] \times 672$ hours or 1680 hours. (The above approximation is based upon an assumption, which is sufficient for a

first order approximation, that life is directly proportional to $1 - S$; where S is the probability of survival.

If the design guide were used to predict life, the following factors would have to be derived:

$$L_A = (D)(E)(F)(G)L_{10}$$

where

- (1) From table 3 [1] for T-1, $D = 0.6$.
- (2) From table 5 [1] for Consumable-Electrode Vacuum-Melting (CVM), $E = 3$.
- (3) Because of the wide variance in hardness of the bearings; that is, Rockwell C 58 to 62, no factor for hardness will be assigned from figure 1 [1].
- (4) From table 10 [1] for an SAE 20 mineral oil at a bearing operating temperature of 100° F with a light series (series 2) bearing, $F = 2$ where the film parameter Λ is 2 (see fig. 2 [1]).
- (5) From table 12, $G = 1$ for a speed of 600 rpm. Therefore,

$$L_A = (0.6)(3)(2)(1)630$$

$$= 2268 \text{ hours}$$

Hence, the field life is approximately 600 hours less than the predicted life using the design guide. Based upon statistical variance and possible misalignment effects which cannot be readily calculated, the predicted life is sufficiently close to that experienced to be used as a design calculation.

What is required for the instant application is a bearing L_{10} life in excess of $\left[\frac{(1-0.90)}{(1-0.94)} \right] \times \left[\frac{(26 \text{ weeks})}{(4 \text{ weeks})} \right] \times 672 \text{ hours}$ or 10 920 hours to assure the probability of there being no bearing failures within a 6 month (26 week) period. This would mean an increase in L_{10} life of a factor of 6 life over field experience. A new bearing material would need to be selected from table 3 [1]. Consumable-electrode vacuum-melted (CVM) AISI 52100 can be chosen. This material has a material factor D of 2 or 3.3 times more than that of CVM AISI T-1. While it is possible to obtain increased life using a heavier series bearing or to specify the controlled differential hardness, (ΔH) concept, the most practical means of further increasing life is to replace the SAE 20 mineral oil with an SAE 30. This would increase the lubrication factor F to 4 which is a twofold increase. The resultant predicted life would be

$$L_A = (2)(3)(4)(1)630$$

$$= 15\,140 \text{ hours}$$

Since field experience has indicated a life approximately 74 percent that of the predicted life L_A , it can reasonably be expected that the L_{10} field life of the bearing will exceed $0.74 \times 15\,140$ hours or 11 203 hours. This life exceeds the requirements of a six month system life.

Example 2

This example is concerned with a gearbox output shaft ball bearing which has experienced at least 5000 hours of maintenance free operation on each of over fifty gearboxes in service. It is desired to increase the output torque of the gearbox by 25 percent and maintain the 5000 hour life of the bearing without redesign of the gearbox. Several gearboxes were put on test at the increased torque condition. The result was a series of bearing failures in a few hundred hours.

The task for the design engineer is to recommend a solution to the bearing problem which will allow the 25-percent torque increase and maintain the 5000 hour minimum life without redesign of the gearbox and, specifically, without changing the envelope dimensions of the bearing.

A description of the bearing and operating conditions are as follows:

215 (light series) deep-groove ball bearing

Ball/race material: Carbon-vacuum degassed (CVD) AISI 52100;
nominal hardness, R_c 62

Surface finish: 8 to 10 microinches CLA on the races; 2 to 3
microinches CLA on balls

Retainer: Machined, iron-silicon-bronze

Lubrication: Single jet

Lubricant: MIL-L-23699

Outer-race temperature: 225° F

Shaft speed: 7200 rpm (inner race rotating)

Thrust load: 480 pounds

Radial load: 600 pounds

The calculated AFBMA life L_{10} for this bearing and conditions is 2180 hours. However, experience has shown that these bearings have run to at least 5000 hours without failure. Use of the Design Guide clearly explains why this is possible. The following life adjustment factors are obtained:

$D = 2$ for AISI 52100
 $E = 1$ for CVD melting process
 $F = 1.5$ for MIL-L-23699 at 225° F;
 $\gamma_o = 40$ SUS and $\Lambda = 1.5$
 $G = 1$ for 7200 rpm
 $H = 1$ assuming misalignment $< 12'$

Then

$$\begin{aligned}
 L_A &= (2)(1)(1.5)(1)(1)2180 \\
 &= 6540 \text{ hours}
 \end{aligned}$$

It can be seen that the 5000 hours without failure could be realized at the initial operating conditions with over 90 percent probability of survival.

The increased torque requirements for the gearbox result in an increase in bearing loads to 600 pounds thrust load and 720 pounds radial load. The calculated AFBMA life at this higher load condition is 1270 hours. Additional heat generation in the gearbox increased the bearing outer-race temperature to 285° F. The increased temperature has great significance in that the lubricant film parameter Λ was reduced to 0.5 due to decreased lubricant viscosity at the higher temperature. [$\Lambda = (\text{lubricant EHD film thickness})/(\text{surface composite roughness}) = h/(\sigma_1^2 + \sigma_2^2)^{1/2}$ where σ_1 and σ_2 are surface finish of balls and races, respectively.] At this condition, excessive surface distress would be expected due to an inadequate lubricant film separating the bearing surfaces. This condition can lead to early superficial pitting failures. According to the Design Guide, (fig. 2), at a Λ of 0.5, the lubrication life factor F could be no greater than 0.2. As a result, the adjusted life

$$\begin{aligned}
 L_A &= (2)(1)(0.2)(1)(1)1270 \\
 &= 500 \text{ hours}
 \end{aligned}$$

This explains the very early failures at the higher torque conditions.

In order to get out of this surface distress condition, Λ must be at least 1.3 for a lubrication-life factor F of 1.0 (Design Guide, fig. 2). Since $\Lambda = m(\mu_o \alpha N)^{0.7} P_o^{-0.09}$, and since $m = 1.8 \times 10^5$, $N^{0.7} = 500$, and $P_o^{-0.09} = 0.55$ are essentially fixed, then

$$\begin{aligned}
 (\mu_o \alpha)^{0.7} &\geq \frac{1.3}{(1.8 \times 10^5)(500)(0.55)} \\
 &\geq 2.62 \times 10^{-8}
 \end{aligned}$$

or

$$\nu_o \geq 38 \text{ SUS}$$

$$\geq 3.6 \text{ cs.}$$

For MIL-L-23699, the operating temperature must be less than or equal to 240° F to attain this 3.6 cs. viscosity. This condition could be attained with additional cooling of the lubricating oil and the bearing.

A kinematic viscosity of about 50 SUS would get away from the very steep portion of the curves in figure 4 of the Design Guide. This would provide more assurance that the surface distress type failure can be averted. At 50 SUS, $(\mu_o \alpha)^{0.7}$ equals 4×10^{-8} and Λ equals 2.0. The bearing operating temperature required for the MIL-L-23699 to operate at 50 SUS (7.4 cs) is 175° F. This temperature is probably too low to expect the oil cooling system to attain considering the increased heat generation at the increased torque condition.

A second alternative is to choose another lubricant with a higher viscosity to satisfy the Λ requirement. Caution should be exercised in the choice of a higher viscosity lubricant, such that excessive frictional losses and heat generation are not caused by too high a lubricant viscosity.

Another alternative is to improve the surface finish of the balls and the race groove to the point where a Λ of 1.3 or greater could be achieved. This fix would require a composite surface roughness $(\sigma_1^2 + \sigma_2^2)^{1/2}$ on the order of 4 microinches CLA by manufacturing the bearing races with a surface finish of approximately 3 microinches CLA. This may not be readily attainable without special race-groove honing techniques.

If the choice is to change to a higher viscosity lubricant or to the improved surface finish without additional cooling, the bearing outer-race temperature is on the order of 285° F. This temperature is in the range where the hardness of AISI 52100 decreases to the extent that life could be decreased. In this case, the recommendation would be to use a material with greater hardness retention at higher temperatures such as AISI M-50. This change would not affect the materials factor D, since D equals 2 for both AISI 52100 and AISI M-50.

Now, assume that the lubrication problem has been solved, and the Λ factor is increased to say 1.3 by one of the above techniques. Then F equals unity. And since D equals 2, E equals unity, G equals unity, and H equals unity remain unchanged, then

$$L_A = (2)(1)(1)(1)(1)1270$$

$$= 2540 \text{ hours}$$

This life is less than the original 5000 hour life which it is desired to maintain. The recommended practice would be to change to consumable-electrode vacuum-melted (CVM) material in lieu of the CVD original specification. The use of the CVM melting process increases the processing variable factor E to 3. The adjusted life is then

$$L_A = (2)(3)(1)(1)(1)1270$$

$$= 7620 \text{ hours}$$

which now should satisfy the original requirement to maintain the 5000 hour life at the increased torque condition and maintain the original bearing envelope dimensions with a 90 percent probability of survival.

Example 3

A continued problem in the design and selection of rolling-element bearings is the inability of the engineer to account for the effect of centrifugal force except by use of a sophisticated computer program. The first two examples above illustrate that rolling-element bearing life generally exceeds the AFBMA life calculation. However, centrifugal force effects can play a significant part in reducing life in many applications. A hypothetical situation would be in the case of a large turbine for electric power generation application having a 236- (180 mm-bore) size roller bearing as one of its components. The basic load rating C of the bearing is determined to be 111 000 pounds based upon AFBMA calculations. In designing the turbine and selecting the bearing, the engineer for the turbine manufacturer considered only bearing catalog life. As a result, for an applied radial load of 11 100 pounds the

$$L_{10} = \left(\frac{C}{P} \right)^n = \left(\frac{111\ 000}{11\ 100} \right)^{10/3} = 2 \times 10^9 \text{ revolutions}$$

The turbine shaft speed is 6660 rpm. Therefore, the expected L_{10} life of the bearing is approximately 5000 hours.

The turbine manufacturer has sold 20 units which are in operation. The turbine manufacturer warrants its unit for trouble free operation for a period of six months or 2500 hours whichever comes first. On a statistical basis, the manufacturer calculates that he may have to replace a bearing in one of the units during the warranty period. However, the manufacturer's engineer assures the company management that based upon his experience, "rolling-element bearings always run beyond the AFBMA calculated life." Shortly after the turbines were placed in the field, the roller bearings began to fail. In a period of three months, the manufacturer had to make good on three warranties and was threatened with a product liability law suit by an irate customer. The immediate reaction of the turbine manufacturer is to complain to the bearing company about "defective bearings." The bearing company immediately pointed out that there were two suppliers of bearings, and that bearings from both suppliers failed. The turbine manufacturer consults its attorney regarding the

probability of collecting damages in a suit against both bearing manufacturers. The attorney employs an independent consulting engineer who submitted the calculations summarized below:

(1) For AISI 52100 steel, the D factor is 2 from table 3 [1].

(2) The material used is carbon vacuum degassed (CVD). No E factor can be assigned to this processing variable.

(3) The hardness of the material was Rockwell C 64 \pm 1 at room temperature, for both the races and rollers. The bearing operating temperature is 300° F. From figure 1, where hardness of the bearings is closely controlled, an E factor for hardness can be obtained where for these bearings E equals 1.5 (see appendix II of [1].)

(4) At 300° F the lubricant, which is a diester oil, has a viscosity of 33 SUS. From the information of table 10 [1], a lubricant factor F was calculated to be 1.6.

(5) The effect of speed was determined from table I [1]. Solving for the DN value of the bearing: 180-mm bore x 6660 rpm = 1.2x10⁶ DN. From figure 9 [1] the speed is within the state-of-the-art.

$$\frac{P}{C} = \frac{11\ 100}{111\ 000} = 0.1$$

or

$$P = 10\% C$$

From table 11(b) for a 180-mm bore cylindrical-roller bearing subjected to a radial load, P equals 10 percent. The speed effect factor G is 0.1.

(6) The misalignment was measured in the bearing on the turbine shaft, and found to be less than 3'. Therefore, H, equals 1.0.

(7) Calculating for L_A

$$\begin{aligned} L_A &= (D)(E)(F)(G)(H)L_{10} \\ &= (2)(1.5)(1.6)(0.1)(1.0)5000 = 2400 \text{ hours} \end{aligned}$$

This life is only one-half the AFBMA life. It would, therefore, not be unreasonable to an engineer competent in bearing design and practice to anticipate approximately three bearing failures within the time frame of three months. As a result, it can be concluded that the bearings supplied by the bearing manufacturers were not defective and were achieving normal lives even though less than those anticipated by the turbine manufacturer.

Example 4

An aircraft engine designer has chosen a 120-mm bore angular-contact ball bearing as a mainshaft bearing for a turbojet engine application. The calculated bearing thrust load during cruise conditions is 1380 pounds. The mainshaft speed is 12 000 rpm. Lubrication is by oil jet. At cruise conditions, bearing outer-race temperature is calculated to be approximately 320° F. The engine manufacturer desires that the bearings have a probability of survival of 98 percent at the 10 000 hour overhaul period.

A lubricant and bearing material must be selected. The engine manufacturer has a computer program which enables an accurate prediction of bearing temperatures, thermal gradients, lubricant in and out temperatures, and prediction of bearing skidding conditions. However, prior to the use of the program, the designer must make some of the basic engineering decisions for a first order analysis. Based upon prior practice, the project engineer selects a bearing with an outer-race riding cage made of silver-plated AISI 4340 steel. He further selects a split inner-race type bearing of ABEC-5 grade having a nominal contact angle of 20°. The reason for the split-inner ring selection is to enable the bearing to take thrust reversal and, hence, thrust in both directions. The inner- and outer-race conformities are selected to be 54 and 52 percent, respectively. (Conformity is the race-groove radius defined as a percentage of the ball diameter where $\text{Conformity} = \left[(\text{race groove radius}) / (\text{ball diameter}) \right] \times 100$.) The bearing contains fifteen 13/16-inch diameter balls. The number of balls and conformity of the inner and outer races were selected on the basis of past operating experience. However, the race geometry and contact angle can be justified analytically on the basis of minimal bearing internal heat generation and minimized contact stresses for an applied load under the specific operating condition enumerated above.

Based upon table 6 [1], the engineer selects MIL-L-9236B as the lubricant. However, the lubricant choice is not always dictated by the designer. An example would be where a specific type lubricant is currently utilized in commercial hardware by the customer. This lubricant would be the one most likely to be chosen. Where this is the case, the engineer would have to design the engine operating conditions to fit the lubricant rather than select the lubricant to fit the operating conditions.

It is well known that AISI M-50 and the other tool steels listed in table 1 [1] more readily retain their hardness at temperatures beyond 250° F than AISI 52100. Further, M-50 is the most commonly used through-hardened steel except for 52100. For these reasons, the engineer selects CVM AISI M-50 for the bearing material. Commercially, M-50 is readily obtainable for bearing rings at Rockwell C hardnesses of 62±1 and for balls at 62±1/2. Actually, with proper control of the carbon content of the material, a one-point Rockwell C hardness increase may be obtainable.

From table 3, for M-50, $D = 2$.

From table 4, for CVM, $E_1 = 3$.

Since bearing hardness is a controlled variable, from figure 1, E_2 equals 1.2.

Solving for the lubricant factor F , as discussed in the previous examples, where the viscosity of the ester lubricant at 320°F is approximately 30 SUS, F equals 1.8.

In order to determine the speed factor G , the bearing DN value is calculated where

$$\text{DN} = (12\ 000\ \text{rpm})(120\ \text{mm}) = 1.44 \times 10^6$$

The basic load rating $C = 14\ 000$ pounds

$$\left(\frac{P}{C} \times 100\right) = \left(\frac{1380}{14\ 000} \times 100\right) = 10\%$$

From table 11(b), $G = 0.9$

It is assumed that there is no misalignment in the bearing.

$$\begin{aligned} L_A &= (2)(3)(1.2)(1.8)(0.9)\left(\frac{14\ 000}{1380}\right)^3 \\ &= 12.6 \times 10^9\ \text{inner-race revolutions} \\ &\text{or } 17\ 500\ \text{hours} \end{aligned}$$

To obtain an estimate of the life at a 98 percent probability of survival,

$$L_2 \approx L_A \left(\frac{1 - 0.98}{1 - 0.90}\right) \approx 3500\ \text{hours}$$

The engineer must conclude that, based upon this first order approximation, he must either select a bearing with a higher load capacity or reduce the design load on the bearing by approximately 30 percent to achieve the reliability required for 10 000 hours between overhaul.

Example 5

In a gear box application, a 310-S (50-mm bore) size roller bearing is utilized. The lubricating oil is an SAE 20 mineral oil with a standard additive package. The bearing inner-race speed is 3000 rpm. The bearing material is CVD AISI 52100. From the manufacturer's catalog, the basic load rating C for this bearing is 17 000 pounds. The bearing radial load is 5000 pounds. The life of the bearing is

$$\begin{aligned} L_{10} &= \left(\frac{17\ 000}{5000}\right)^{10/3} \\ &= 59 \times 10^6\ \text{inner-race revolutions} \\ &\text{or } 328\ \text{hours} \end{aligned}$$

Since this gear box is not in continuous operation, the bearing life is theoretically adequate for the application. However, because of a thrust load which was not anticipated nor could be calculated in the initial design of the gear box, roller end wear of the bearing has become a serious problem necessitating premature gear box overhaul. The engineer for the gear box manufacturer decides that the use of a deep-groove ball bearing or a tapered roller bearing would solve the thrust problem.

Utilizing manufacturer's catalogues, the project engineer further decides that 310-S (50-mm bore) size deep groove ball bearing or a 455 (50-mm bore) series tapered roller bearing can fit the current gear housing dimensions without significant modification to the gear box. The project engineer assigns to one of his design engineers the responsibility of choosing between the ball or tapered bearing for the application.

The design engineer who normally has gone to a specific manufacturer's catalog to select bearings for a specific application finds that this manufacturer does not produce tapered roller bearings. The engineer finds that there are no readily available texts to assist him in his problem. He consults the AFBMA standards for tapered roller bearings and finds them nonexistent. As a result, he turns to a manufacturer of tapered roller bearings from whom he receives the following information.

The bearing basic load rating (C in the AFBMA standard) is referred to as either BRR (basic radial rating) or BTR (basic thrust rating) for tapered roller bearings. This is the load which will fail the bearing in 90×10^6 inner-race revolutions. (The definition of BRR and BTR may be different with different manufacturers.) The engineer recalls that C is the load which will fail the bearing in one-million revolutions according to the AFBMA definition. The BRR for this tapered roller bearing is 5400 pounds. Therefore, the catalog life of the bearing is

$$L_{10} = 90 \left(\frac{\text{BRR}}{P} \right)^{10/3} = 90 \left(\frac{5400}{5000} \right)^{10/3}$$

$$= 116 \times 10^6 \text{ inner-race revolutions}$$

$$\text{or } 645 \text{ hours}$$

(The thrust load was considered to be extremely small relative to the radial load for purposes of the above calculation.)

For the deep-groove ball bearing, C equals 10 800 pounds. Therefore, for the current application

$$L_{10} = \left(\frac{10\,800}{5000} \right)^3$$

$$= 10 \times 10^6 \text{ inner-race revolutions}$$

$$\text{or } 56 \text{ hours}$$

It becomes obvious to the engineer that in this application, the tapered bearing will produce the longest life. However, the

project engineer requests that consideration be given to material selection and lubrication effects. Using the Design Guide [1], the engineer calculates Λ for the tapered bearing to be approximately 3 for an SAE 20 oil at 130° F (~160 SUS viscosity) outer-race temperature and a medium series bearing. This produced a lubricant factor F from figure 2 [1] of 2.4. The bearing material is a case carburized air melted AISI 8620 of a minimum Rockwell C hardness of 58. For this material, both Material Factor D and Processing Factor E are 1. The Speed Effect Factor G can be determined by converting the BRR to C. Dividing the radial load by C will allow use of table 11(b). For the instant application P/C is greater than 20 percent. Hence, G equals unity at 0.25×10^6 DN. The life of the tapered bearing

$$L_A = (1)(1)(2.4)(1)(645)$$

$$= 1620 \text{ hours}$$

A similar calculation for the plain roller bearing previously used would have yield the following for air-melt AISI 52100 during a D factor of 2:

$$L_A = (2)(1)(2.4)(1)(328)$$

$$= 1570 \text{ hours}$$

Hence, while the fatigue lives of the plain roller and tapered bearings are similar, the tapered bearing would have the ability to withstand the system thrust load. For the ball bearing, it would have been feasible to change the bearing material to CVM AISI 52100. The life L_A of the bearing would be approximately 800 hours. As a result, the project engineers selection of whether to use the ball bearing or tapered bearing probably would be an economic one since each bearing would achieve the gear box design life.

Example 6

Not all applications of ball and roller bearing life predictions need be for design purposes. The economic necessity of predicting life also is important to test laboratory evaluations of bearings. In many cases where there is a new bearing design, material, or lubricant to be evaluated, the project engineer must be able to predict life to a reasonable degree of accuracy for purposes of costing his program or of determining the amount of test stand time required to assure meaningful results. This time prediction becomes even more important when the laboratory is in a competitive bid to provide services. An estimation of life which is excessive can result in the laboratory losing the contract. On the other side of the ledger, an estimate of time less than that required for testing can result in a loss to the laboratory or being disqualified from consideration for providing the required services. The following is an example of the application of the design guide for the purposes of estimating test stand hours.

A user of bearings for automobile applications desires to determine the experimental life of the bearings supplied by three bearing manufacturers. The user would like to compare the bearings with the AFBMA calculated life which its engineers used in their design. The actual bearing life is important because of warranty considerations. The bearing has the following specifications:

Size 113 (65-mm bore)
 Ball diameter, 0.4375 inch (11.112 mm)
 Pitch diameter, 3.248 inch (82.5 mm)
 Number of balls, 14
 Inner-race diameter, 2.811 inch (71.388 mm)
 Outer-race diameter, 3.686 inch (93.61 mm)
 Race conformity, 52 percent
 Material, CVD AISI 52100
 Rockwell C hardness, 63±1

The test laboratory sets specific limitations on the test conditions. First of all, the maximum contact (Hertz) stress on the bearing during test should not exceed 350 000 psi even though under actual operation the stress is 200 000 psi. The bearing temperature should not exceed 180° F. The lubricant to be used is an SAE 10 mineral oil at 180° F. From this criteria, the engineer performs the following calculations using [4].

Principal radii of curvature:

Ball,

$$r_{1I} = r_{2I} = \frac{11.112}{2} = 5.556 \text{ mm}$$

Inner Race,

$$r_{1II} = 5.778 \text{ mm}$$

$$r_{2II} = \frac{71.388}{2} = 35.694 \text{ mm}$$

$$R_{1I} = \frac{1}{r_{1I}} = 0.1800; \quad R_{2I} = \frac{1}{r_{2I}} = 0.1800$$

$$R_{1II} = \frac{1}{r_{1II}} = -0.1730; \quad R_{2II} = \frac{1}{r_{2II}} = 0.02801$$

$$P_o = \left(\frac{S_{\max} \pi e_a e_b \epsilon_E^2}{1.5} \right)^3 \left(\frac{1}{\sum R} \right)^2$$

$$S_{\max} = \left(\frac{3.5 \times 10^5 \text{ lb.}}{\text{in}^2} \right) \left(\frac{\text{in.}}{25.4 \text{ mm}} \right)^2 \left(\frac{\text{Kg}}{2.205 \text{ lb}} \right)$$

$$= 246 \text{ Kg/mm}$$

From [4]

$$e_a, e_b, \epsilon_E = F(R)$$

where

$$F(R) = \frac{(R_1 - R_2)_I + (R_1 - R_2)_{II}}{(R_{1I} + R_{2I} + R_{1II} + R_{2II})}$$

From table 2.1 [4]

$$e_a = 0.1885, e_b = 0.0212, \epsilon_E = 1.00$$

Hence

$$P_o = \left(\frac{246 \times 3.14 \times 0.1885 \times 0.0212 \times 1}{1.5} \right)^3 \left(\frac{1}{0.2150} \right)^2$$

$$= 189 \text{ Kg} = 416 \text{ lb}$$

This is the load on the heaviest loaded ball in the bearing. The following calculation converts to the bearing radial load where

$$P = \frac{P_o(\text{number of balls})}{4.37} = \frac{(416)(14)}{4.37}$$

$$= 1332 \text{ lb}$$

From the AFBMA standard for this bearing C equals 5840 pounds.

Therefore,

$$L_{10} = \left(\frac{C}{P} \right)^3 = \left(\frac{5840}{1332} \right)^3$$

$$= 84 \times 10^6 \text{ inner-race revolutions}$$

$$\text{or } 140 \text{ hours at a test speed of } 10\,000 \text{ rpm.}$$

From the Design Guide [1], the following factors are determined:

$$D = 2 \text{ (table 3)}$$

$$E_1 = 1 \text{ (table 5)}$$

$$E_2 = 1.43 \text{ (fig. 1)}$$

$$F = 2.8 \text{ (table 10)}$$

$$G = 1 \text{ (fig. 8)}$$

$$L_A = (2)(1)(1.43)(2.8)(1)(140)$$

$$= 1120 \text{ hours}$$

From the above, it must be concluded that the test conditions are too restrictive to obtain fatigue data within a reasonable time and at reasonable costs. Were the test laboratory to quote a fixed price based upon the AFBMA calculations, a significant financial loss to the laboratory would have occurred. Changing the test temperature from 180° F to 210° F would decrease L_A to 470 hours by reducing lubricant factor F to 1.2. Based upon a 470 hour L_A and using the methods of [3], it can be calculated that the number of test hours for each of the three series based upon 40 test bearings with 10 failures per series are approximately 51 thousand bearing test hours.

Concluding Remarks

The examples given in the text are meant to bridge the gap between the ASME Design Guide and actual engineering practice. The problems were selected to be representative of typical problems which may be encountered by the engineer in design. However, the problems themselves are not meant to be overall encompassing to preclude the engineer from using more exact means of analysis such as computer programs which factor the various variables to predict not only bearing life, but temperature gradients, thermal growth, skidding of the rolling elements, elastohydrodynamic film thickness at each rolling-element contact, bearing torque, and centrifugal effects.

The engineer must also realize that lubricant filtration can play an important part in bearing life. The failure to filter out wear particles which are several times greater than the lubricant film thickness can significantly decrease life from that calculated. Bearing installation also is an important factor which if not carefully controlled can induce misalignment in the bearing. Corrosion is another important variable which can be induced by failure to place lubricant on a bearing subsequent to cleaning and prior to installation or during periodic maintenance. Another common problem related to bearing failure is lubricant starvation either because of the failure to replenish the lubricant sump with oil or because of dirt contamination in the system. However, if all the above factors are properly controlled, the calculated bearing fatigue life will be the limiting time for successful bearing operation with a 90 percent probability of survival.

REFERENCES

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4. Palmgren, A., Ball and Roller Bearing Engineering, 3rd ed., SKF Industries, Philadelphia, Pa., 1959.